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N63-22713

NASA TECHNICAL NOTE

NASA TN D-1994

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MINIMUM-OIL-FLOW REQUIREMENTS OF HIGH-SPEED BALL BEARINGS AT TEMPERATURES TO 800° F

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • OCTOBER 1963

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SUMMARY

A total of 58 minimum-oil-flow tests were run on eight pairs of 75-millimeter-bore 215-size M-2 tool-steel deep-groove ball bearings at DN values (product of bearing bore in mm and speed in rpm) ranging from 0.45×10^6 to 1.2×10^6 and thrust loads of 1000, 2000, and 3000 pounds. The bearings were air-oil-mist lubricated with highly refined naphthenic mineral oil (MLO 7243). Tests were run under two general conditions: (1) with no external heat flow to the bearings and (2) with external heat added to maintain constant bearing temperatures of 400°, 500°, 600°, 700°, and 800° F.

Minimum required oil flow for continuous bearing operation increased with increasing DN and load.

Bearings operated satisfactorily at very low oil flows with an air-oil-mist system of lubrication. For example, it was possible to operate 215-size bearings satisfactorily at 8000 rpm (DN of 0.6×10^6), 800° F, and a 1000-pound thrust load with an oil flow of 5.5×10^{-3} pound per minute.

Minimum required oil flow increased with increasing bearing temperature up to 500° F. At 600° F, the minimum required oil flows were less than at 500° F, apparently because of the beneficial effects of the varnish deposits by the decomposing lubricant on the bearing parts. Bearing temperature and torque generally decreased with decreasing oil flow as the minimum oil flow was approached.

INTRODUCTION

Many bearings in supersonic aircraft and space vehicles will be required to operate at high speeds and temperatures approaching 1000° F. The usefulness of organic liquids as lubricants is limited by their oxidative and thermal stability. The extent of oxidation of a lubricant is determined by the temperature and the time at temperature if there is sufficient oxygen available for the reaction to take place. Oxidation will not occur if oxygen is absent, but thermal degradation, which is also a function of time and temperature, will take place in any atmosphere. Lubricants are thermally stable to temperatures higher than the temperature at which they readily oxidize so that their useful temperature range can be extended by limiting the available oxygen (ref. 1).

Since thermal and oxidative degeneration are functions of time and temperature, it would appear to be advantageous in some high-temperature applications to use a "once-through" lubrication system. The lubricant would be exposed to the high temperature for the shortest possible time so that degradation would be minimized and the lubricant would then be discarded.

In conventional recirculating lubrication systems, flow rates of the order of gallons per minute are employed. Most of the lubricant serves as a coolant, which makes it necessary to provide a heat exchanger to remove heat from the lubricant. Cooling capacities become more and more limited with increasing flight speeds because of aerodynamic heating. Bearing temperatures will be governed to some extent by the available cooling capacity. A once-through lubrication system reduces the heat-sink requirement because only enough lubricant is supplied to provide the lubricating function; thus extremely low oil flows are permitted. Further, a once-through system requires less accessory space and weight than a recirculating system. Its use also permits consideration of fluids that would not be satisfactory in a recirculating system. Equilibrium bearing temperatures in a once-through system will be higher than in a recirculating system. The disadvantage of higher bearing temperatures in terms of bearing-material problems in a once-through system must be weighed against the problem of cumulative lubricant degradation in a recirculating system. In the temperature range 500° to 1200° F, bearing-material problems do not appear to be too severe (refs. 2 to 5).

The once-through system, therefore, appears to have a definite advantage for relatively short flights. Very little experimental information exists on the minimum-oil-flow requirements of rolling-contact bearings. In reference 6, it is reported that a 215-size ball bearing operated satisfactorily for 6.64 hours on 1 cubic inch of a diester lubricant (an oil flow rate of 3×10^{-3} lb/min) at 12,000 rpm, a 1000-pound thrust load, and 400° F. The data of reference 6 indicate that high-speed bearing systems can be operated for considerable periods of time on a very nominal quantity of lubricant at temperatures to 500° F.

References 7 and 8 report some data on the relation of the running time to failure as a function of the quantity of oil applied prior to a test for 30- and 50-millimeter-bore ball bearings. In reference 8, the minimum oil quantity is deduced on a steady-flow basis from the analysis of experimental data; this information was obtained to secure quantitative estimates of the required bleed rate of greases.

Reference 9 reports some data on minimum-oil-flow requirements of 25-millimeter-bore ball bearings at a DN (product of bearing bore in mm and speed in rpm) of 0.8×10^{6} and operating temperatures of ball and roller bearings at low oil flows.

To design a once-through lubrication system properly, a knowledge of the oil-flow requirements of a bearing at various operating conditions is required. The prime objective of this investigation, therefore, was to determine the performance limitations and actual flow requirements of ball bearings over a range of speeds, loads, and temperatures. These experiments were conducted at DN values of 0.45×10^{6} to 1.2×10^{6}, thrust loads of 1000, 2000, and 3000 pounds, and temperatures to 800° F. The approximate mean Hertz stresses varied from 149,000 to 228,000 pounds per square inch. The test lubricant was a highly refined naphthenic mineral oil that was designated as either MLO 7243 or MLO 7277 (ref. 10). This lubricant was selected from nine high-temperature lubricants evaluated in tests conducted on 204-size ball bearings at temperatures of 500°, 600°, and 700° F. Low bearing torque and moderate deposits from the naphthenic mineral oil over the temperature range of these tests made this lubricant the best selection. The lubricant was supplied to the test bearings as an air-oil mist.

APPARATUS

Test Rig

A cutaway view and cross sections of the test-bearing arrangement are shown in figure 1. With the exception of the test-bearing arrangement, the rig is basically the same as that used in reference 6. The two test bearings were loaded in thrust by a flexible stainless-steel bellows, which was pressurized with nitrogen gas (fig. 1(b)). The bellows was calibrated on a tension-compression test machine. The load capacity of the system was about 5000 pounds. The available speed range of the test shaft was 1100 to 36,000 rpm, controllable to within ±50 rpm at all speeds.

The test shaft was cantilever mounted and supported by two deep-groove ball bearings. The bearings were lubricated by a full-flow recirculating system with a MIL-L-7808C diester oil. A water-cooled carbon face seal (fig. 1(b)) was used to prevent the support-bearing oil from entering the test-bearing chamber.

A strain-gage beam connected to the bearing housing (fig. 1(a)) was used to measure the friction torque of both test bearings. The strain-gage output was amplified and recorded on a photoelectric potentiometer. Torque could be measured with an average sensitivity of 0.05 inch-pound. An electronic torque-limiting device automatically decreased the speed and load and increased the oil flow when the torque of the two test bearings exceeded 10 inch-pounds.

Temperature Measurement and Control

Induction heating coils were wound in a recess in each bearing housing. Two iron-constantan thermocouples were located in each test-bearing outer-race housing 180° apart at the axial midpoints of the bearings. These thermocouples were embedded in the housing flush with the housing inside diameter. The thermocouples were connected to a two-pen strip-chart recorder, which recorded the individual test-bearing outer-race temperatures.

The test-bearing temperature was controlled by two methods. The test bearings were heated with an induction heating unit, and temperature was controlled within ±10° F by varying the power input to the induction heater. To maintain both test bearings at approximately the same temperature, air at room temperature could be introduced through the annular cooling rings (fig. 1(b)). A number of orifices in each cooling ring directed jets of air at each test bearing.

Oxygen Measurement

To determine the oxygen content of the bearing atmosphere, a paramagnetic oxygen analyzer was used. This unit continuously monitored the oxygen content of the gas emerging from the front test bearing. The gas sample was drawn from a probe positioned 1/8 inch from the retainer of the front test bearing to the oxygen analyzer. The output of the analyzer was connected to a strip-chart recorder, which continuously recorded the instantaneous oxygen content. The oxygen content was determined with an average measured sensitivity of 0.125 percent in the range 0 to 25 percent.

Test-Bearing Lubrication System

In order to obtain the range of oil flows required for this investigation, an air-oil-mist lubricator was designed. A sketch of this system is shown in figure 2. The experimental lubricator employed a pressurized tank and a capillary to feed lubricant into the flow line downstream of a Venturi. The nozzle that directed the oil to each test bearing acted as a reclassifier in that it increased the oil particle size to cause satisfactory wetting of the test-bearing surfaces. As in reference 6, several combinations of jets and capillaries were used to obtain the desired range of flow of test lubricant.

Different combinations of capillaries and jets were calibrated at various supply pressures with test tubes partly filled with glass wool to trap the oil flow. The available oil-flow range with all combinations of capillaries and jets was 5×10^{-5} to 0.1 pound per minute.

The lubricant used in these tests was a highly refined naphthenic mineral oil whose properties are given in table I.

Test Bearings

All tests were performed with 215-size deep-groove M-2 tool-steel bearings with two-piece, outer-race-riding, annealed M-2 retainers. The test bearings had outer- and inner-race conformities of $f_o = 0.53$ and $f_i = 0.53$ and were ABEC 5 grade. The nominal percentage composition of M-2 tool steel is as follows: carbon, 0.80; chromium, 4.0; vanadium, 2.0; molybdenum, 5.0; tungsten, 6.0; manganese, 0.30; and silicon, 0.30. The test bearings had an average radial clearance of 0.0014 inch, with a range of radial clearance from 0.0012 to 0.0016 inch. The average hardness of the test races and balls was Rockwell C-63.

PROCEDURE

Test Procedure

Minimum-oil-flow data were obtained in the following manner. After the desired load, speed, and temperature were selected, a flow that from previous experience was known to be sufficiently high to ensure proper bearing operation under the chosen conditions was set. The test bearings were run at this flow for 1 hour to ensure that equilibrium conditions prevailed. The flow was then halved, all other conditions remaining the same, and another 1-hour run was attempted. This procedure was continued until an incipient failure occurred; failure consisted of a sudden sharp rise in test-bearing torque (fig. 3) exceeding 10 inch-pounds, or a 50° F temperature rise. Momentary inadequate lubrication due to a local breakdown of the boundary lubricating film without significant surface damage is hypothesized. The speed and load were decreased, and oil flow was increased by the electronic torque-limiting device when the torque exceeded 10 inch-pounds. The bearings were then operated at an idling speed of 5000 rpm for 1/2 hour to allow the bearings to recover. Recovery was indicated by lower and more stable bearing torque and temperature. Frequent checks of previous data were made throughout the test program to make certain that bearing performance was not affected by the incipient failures. When the bearing characteristics changed by a factor greater than the expected experimental error, the test bearings were replaced.

The DN range of this investigation was limited by the range of oil flows obtainable with the air-oil-mist lubrication system. The minimum oil flows required to accomplish this investigation varied from 5×10^{-5} to 0.1 pound per minute.

Types of Tests

Two basic types of tests were conducted. Five tests were run with no external heat flow to the bearings. In these tests, the bearing temperature was allowed to seek an equilibrium level and was a function of both DN and load. Fifty-three tests were run with the test bearing outer-race temperature maintained approximately constant by variance of the heat flow into the bearing housing. Tests of this type were run at bearing temperatures of 400°, 500°, 600°, 700°, and 800° F.

RESULTS AND DISCUSSION

The results of this experimental investigation are presented in figures 4 to 9. The effects of DN, load, and temperature on minimum-oil-flow requirements and bearing torque will be discussed separately. Oxygen concentration in the bearing atmosphere was continuously monitored and was used to explain some of the results.

Effect of DN

The effect of DN on the minimum required oil flow at various temperatures and a 1000-pound thrust load is shown in figure 4. The rate of change of minimum oil flow is constant with DN at a constant temperature when semilog coordinates are used. The slopes of the minimum-oil-flow curves at constant temperature are approximately the same except for the 700° and 800° F curves, which are somewhat steeper. Therefore, to determine the change in oil-flow requirements due to a change in DN in the temperature range 400° to 600° F, only a minimum oil flow at a reference DN value and the slope are needed. The following approximate equation may be written for minimum required oil flow Q as a function of DN, which is valid in the temperature range 400° to 600° F:

$$Q = e^{6.2DN'-A}$$

where $DN' = DN \times 10^{-6}$ and A has the following values:

Temperature, °F	A
400	10.91
500	8.30
600	15.71

A similar equation may be written for the 700° and 800° F lines. As DN is doubled in the temperature range 400° to 600° F with load and temperature constant, the minimum-oil-flow requirement is increased by a factor of approximately 50 for DN values between 0.6×10^6 and 1.2×10^6. As an example of the quantity of oil required, figure 4 shows that an oil flow of 5.5×10^{-3} pound per minute is adequate for satisfactory bearing operation at 8000 rpm (DN of 0.6×10^6) and 800° F. This is equivalent to operation of the bearing for more than 3 hours on 1 pound of lubricant.

Note that the slope of the curve for no heat addition in figure 4 is not constant and that the slope of this line is considerably steeper than the constant-temperature lines at DN values up to 1×10^6. This is so because two factors, DN and temperature, determine the minimum required oil flow when the bearing temperature is not controlled. As the DN value is increased with no heat addition to the bearing, the bearing free-running temperature also increases. The rise in bearing temperature with DN is responsible for the high initial slope of the no-heat-added curve in figure 4. The rate of temperature rise decreases as the DN value is increased. This explains why the slope of the no-heat-added curve decreases at higher DN values.

Effect of Load

The effect of load on the minimum required oil flow is shown in figure 5(a) for a bearing temperature of 400° F and in figure 5(b) for a bearing temperature of 700° F. From figure 5, it is apparent that an increase in the load results in an increase in minimum required oil flow. Comparison of figures 5(a) and (b) reveals that the minimum oil flow is more sensitive to load at 700° F. At 700° F the bearing requires a minimum oil flow almost 20 times as great at 2000 pounds as at 1000 pounds. At 400° F, however, the bearing will operate satisfactorily with a 2000-pound load and a minimum oil flow just two to four times that required with a 1000-pound load.

The curves of figure 5(a), which increase in slope as the load increases, show that minimum required oil flow is more sensitive to DN at higher loads. The minimum required oil flow obtained in these tests probably represents the minimum flow necessary to maintain satisfactory boundary lubrication between the bearing parts in rolling and sliding contact. Evidently, the effects of load and sliding velocity (proportional to DN) on the minimum required oil flow are compounded so that the sensitivity of minimum oil flow to either of these variables is greater at high values of the other.

Effect of Temperature

Figure 6 shows the effect of temperature on minimum-oil-flow requirements. The minimum flow requirements increase with temperature at bearing temperatures to 500° F. In the investigation of reference 2, the minimum flow requirements for a diester oil increased with temperature at bearing temperatures to 400° F. These results are consistent with the postulation that the minimum required oil flow represents the minimum flow necessary to maintain satisfactory boundary lubrication because the rate of evaporation increases with temperature.

At 600° F, however, the test bearings operated satisfactorily at oil flows approximately 1/80 of those required at 500° F. They continued to operate satisfactorily at this low oil flow until the bearing temperature reached approximately 625° F. Attempts to determine required minimum oil flows at temperatures between 500° and 600° F and between 625° and 700° F were unsuccessful because of erratic operation in these temperature regions. The curves of figure 6 are drawn as dashed lines between 500° and 600° F and between 600° and 700° F because their shapes in these temperature regions are undefined.

It is known that oxidative and thermal degradation of the lubricant at elevated temperatures results in an accumulation of deposits on the hot surfaces. These deposits act as a secondary solid lubricant that helps to maintain satisfactory boundary lubrication (refs. 6 and 11 to 13).

Deposition tests were performed with the test lubricant in an effort to learn more about its deposit-forming characteristics at high temperatures. The apparatus used to perform these tests is described in reference 14. The effect of temperature on deposition, viscosity, and acid number at airflow rates of 1000, 2000, and 3000 cubic centimeters per minute is shown in figure 7. The results at 2000 and 3000 cubic centimeters per minute appear to indicate that the deposition rate reaches a maximum at 650° F. At temperatures above 650° F, peeling was observed, which indicated that some of the deposition products were not adhering to the tube surface. Maximum adherence was obtained at 650° F. It should be recalled that test-bearing temperature is measured on the outer-race periphery, and perhaps the ball track temperature could be approximately 650° F or greater for the 600° F bearing temperature. It is postulated that, in these experiments, the optimum deposition conditions occurred at a measured bearing temperature of approximately 600° F. These deposits could explain the minimum-oil-flow requirements in the 600° F temperature region. The apparently beneficial nature of the deposits increases the number of lubricants that may be used in a once-through system, many of which would be eliminated for use in recirculating systems.

Figure 7 also shows the variations in viscosity and acid number with temperature and airflow. Viscosity increases with both temperature and airflow. This is to be expected since increasing the temperature and the available oxygen results in higher rates of oxidative and thermal breakdown of the lubricant and more viscous degradation products are formed.

The acid number increases with airflow and temperature until a maximum acidity is reached at a temperature that depends on airflow. The decrease in acidity at higher temperatures probably results from volatile acids being driven off at increasing rates as the temperature is raised.

Further work is necessary to establish conclusively the factors of importance underlying the successful bearing operation with extremely low oil-flow rates in the 600° F temperature region.

It was observed in the tests with no heat addition that bearing temperature decreased with decreasing oil flow (fig. 8(a)). This effect has been observed at low oil-flow rates in other tests (ref. 15). In the low-oil-flow range, bearing temperature decreases as the oil flow is reduced because the power loss or heat generation due to viscous shearing decreases. This is true as long as the bearing surfaces are adequately wetted and good boundary lubrication is maintained. This decrease in power loss or heat generation was verified by measurements of bearing torque in the same oil-flow range (fig. 8(b)). A significant reduction in bearing torque was noted as the oil flow was reduced from 0.0013 to 0.00016 pound per minute. This effect is also reported in reference 6.

Oxygen Content of Bearing Atmosphere

The weight fraction of oxygen in the bearing atmosphere was measured so that the degree of oxidation of the lubricant could be observed. The results are shown in figure 9. The first detection of oxidation of the oil occurred at a bearing temperature of 580° F, when the oxygen content declined 0.125 percent. This was coincident with high, erratic torque. The continued decrease in oxygen content as temperatures were increased indicated an increased rate of oxidation.

SUMMARY OF RESULTS

A series of minimum-oil-flow tests, conducted with 215-size deep-groove ball bearings at DN values ranging from 0.45×10^6 to 1.2×10^6, thrust loads from 1000 to 3000 pounds, and temperatures from 400° to 800° F revealed the following:

1. Ball bearings will operate satisfactorily at very low flows with an air-oil-mist, once-through lubrication system. It is possible to operate a bearing at 800° F, a 1000-pound thrust load, and 8000 rpm (DN of 0.6×10^6) with an oil flow of 5.5×10^{-3} pound per minute. This is equivalent to operation of the bear-

ing for more than 3 hours on 1 pound of lubricant.

2. Minimum-oil-flow requirements for continuous bearing operation increased with increasing DN value and also with increasing bearing load. When the DN value was doubled (0.6×10^6 to 1.2×10^6), the minimum required oil flow was increased by a factor of approximately 50. Also, when the load was doubled (1000 to 2000 lb), the minimum required oil flow at 400° F was increased by a factor of approximately 2 to 4.

3. Minimum-oil-flow requirements increased with increasing bearing temperature up to 500° F. The flow requirements at 600° F were below those at 500° F. At bearing temperatures above 600° F, oil-flow requirements again increased with increasing bearing temperature. It is postulated that the decrease in required oil flow at 600° F was caused by a combination of partial oxidation and thermal degradation of the lubricant. This resulted in a rapid increase in the rate of formation of deposits on the bearing. These deposits are believed to be beneficial as boundary lubricants. In a recirculating system, however, the deposits would be harmful and might make this test lubricant unsatisfactory. On the other hand, the beneficial nature of the deposits increases the number of lubricants that may be used in this type of system, many of which were once eliminated for use in recirculating systems.

4. A slight decrease in measured oxygen concentration occurred around the test bearing when the test-bearing temperature approached 600° F.

5. In the minimum-oil-flow range, bearing temperature and torque generally decreased as oil flow was decreased. This was probably due to a reduction in heat generation caused by a decrease in oil shearing.

Lewis Research Center
National Aeronautics and Space Administration
Cleveland, Ohio, July 22, 1963

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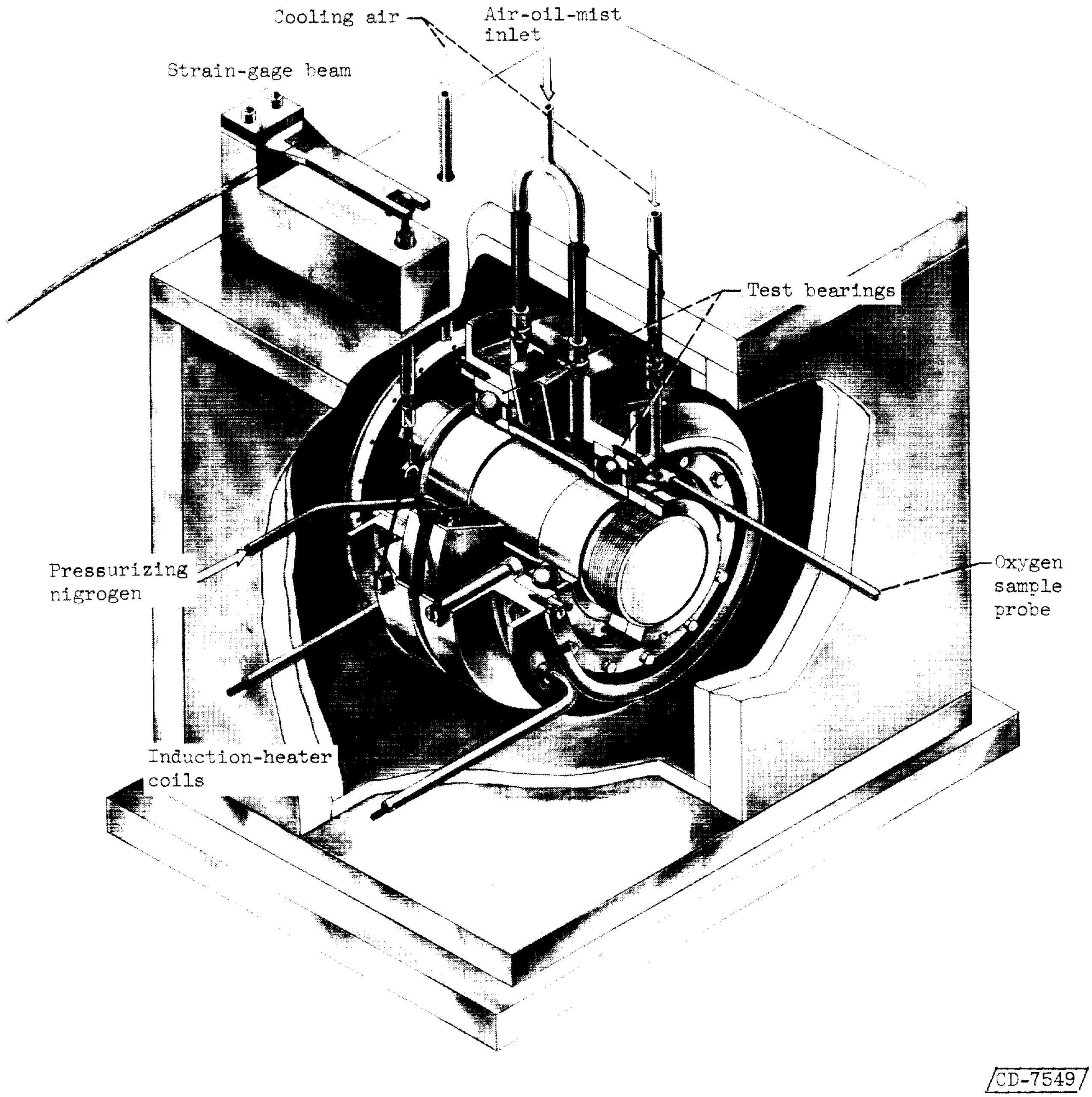
TABLE I. - PHYSICAL PROPERTIES OF

TEST LUBRICANT, MLO 7243

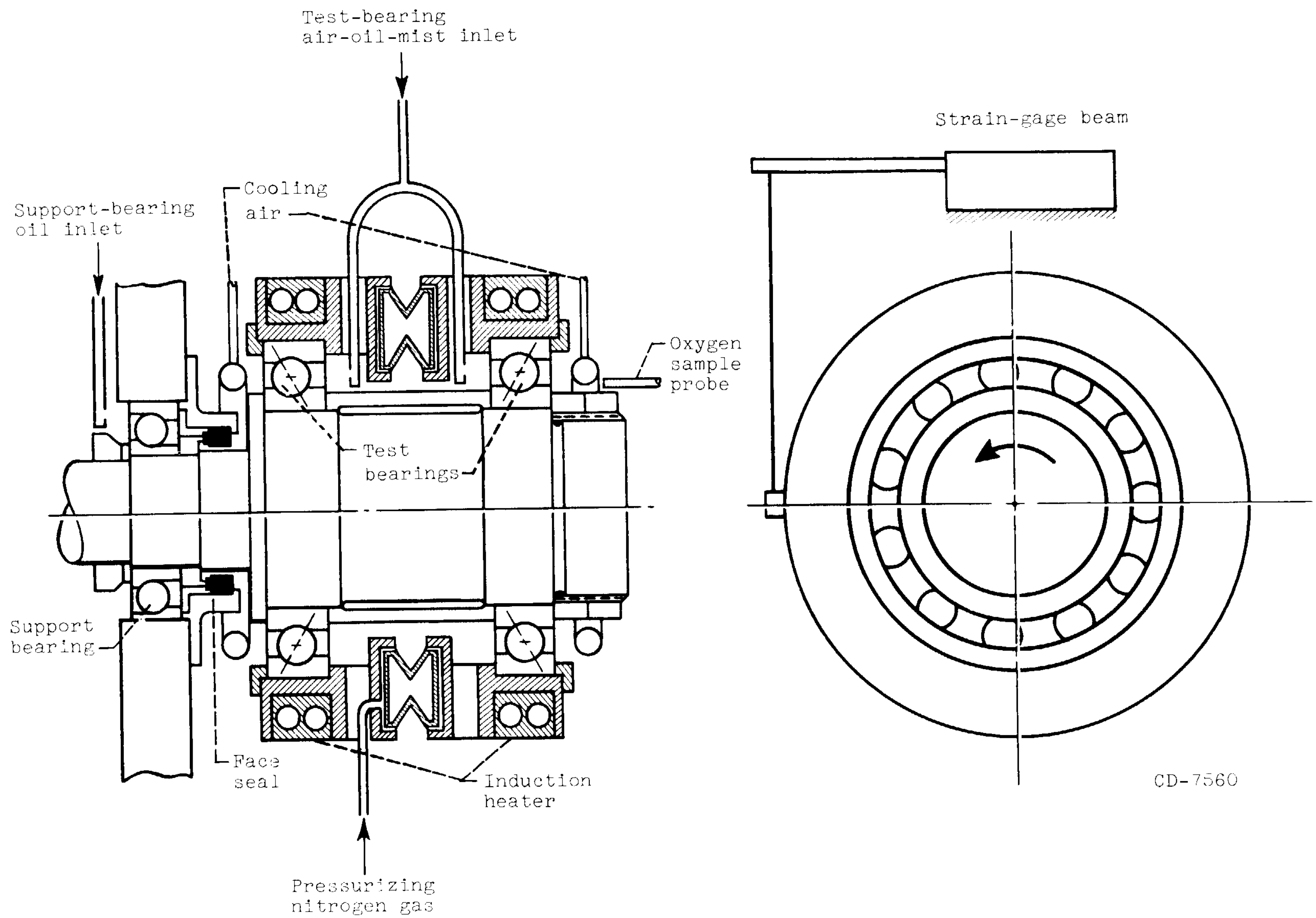
[Data from ref. 10, p. 41.]

Density, g/ml	
0° F	0.908
100° F	.873
200° F	.838
300° F	.802
400° F	.768
500° F	.732
Vapor pressure, mm Hg	
300° F	<0.1
400° F	2.0
500° F	17.0
Viscosity, centistokes	
0° F	10,000
30° F	1,500
100° F	79
210° F	8.4
500° F	1.1
700° F	0.6
ASTM slope (210° to 100° F)	0.759
Viscosity index	80
C.O.C. flash point, $^{\circ}$F	445
C.O.C. fire point, $^{\circ}$F	495
ASTM pour point, $^{\circ}$F	-30



(a) Cutaway view.

Figure 1. - Apparatus for minimum-oil-flow tests.



(b) Cross section.

Figure 1. - Concluded. Apparatus for minimum-oil-flow tests.

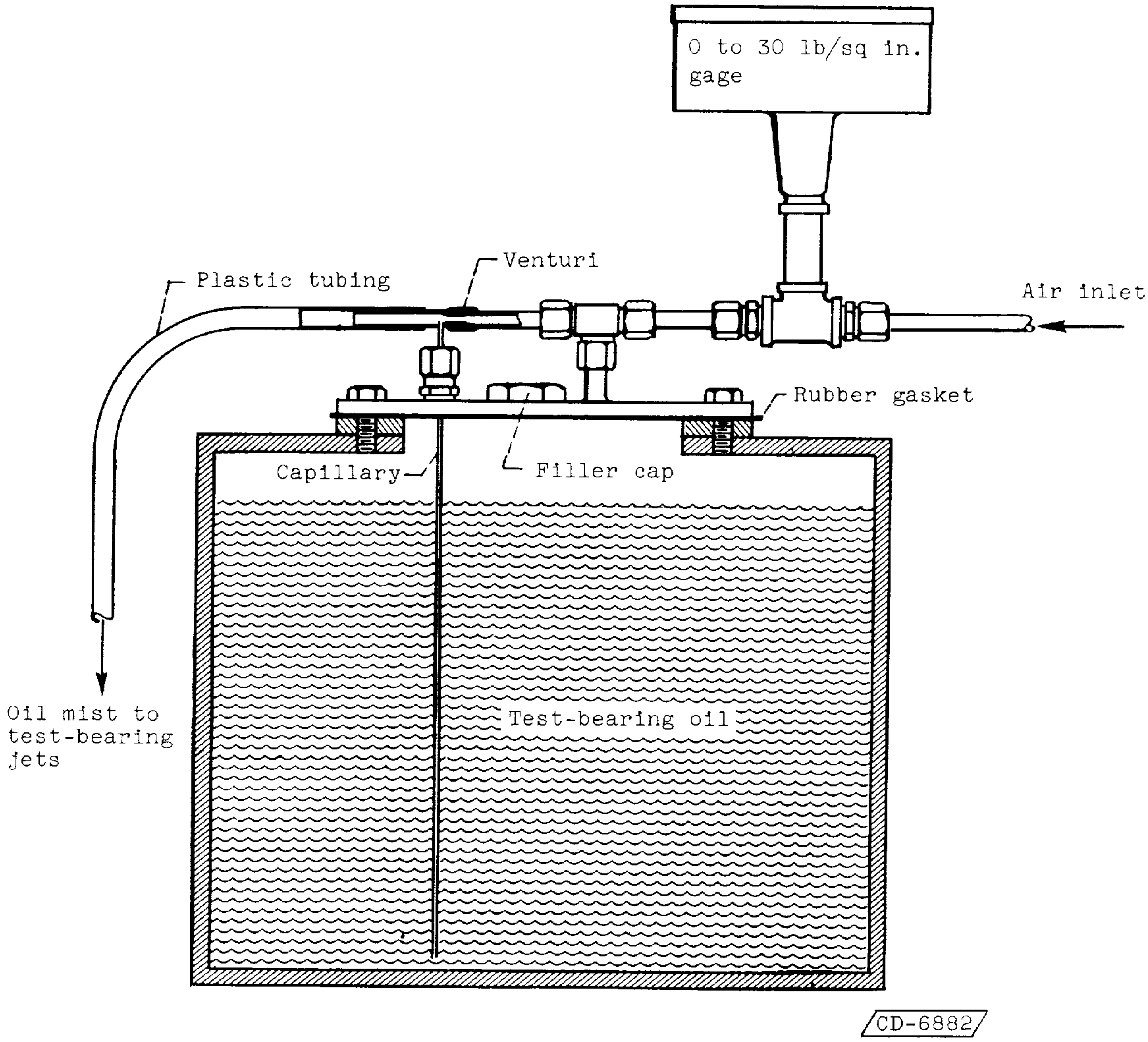


Figure 2. - Experimental air-oil-mist test-bearing lubrication system.

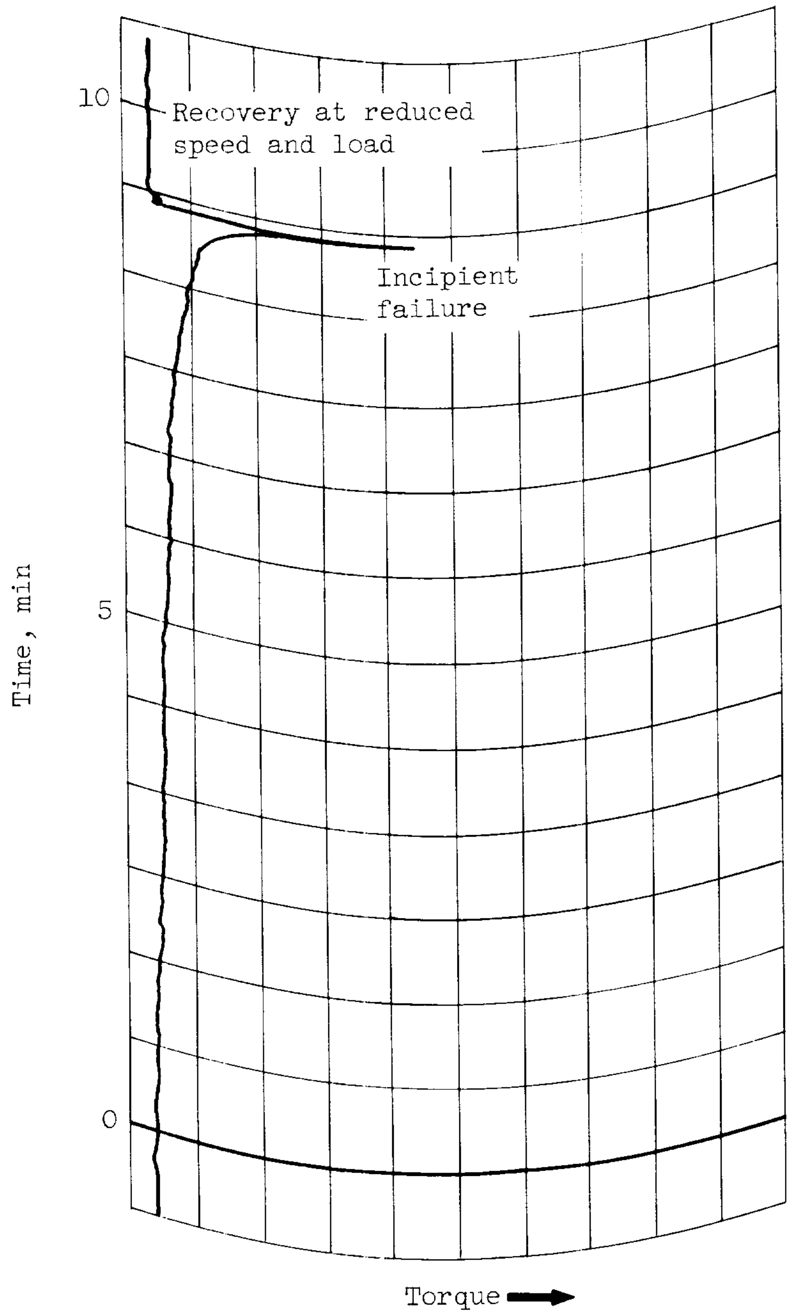


Figure 3. - Typical bearing torque curve at incipient failure. DN, 0.6×10^6; thrust load, 1000 pounds; temperature, 400° F; flow per bearing, 5×10^{-4} pound per minute.

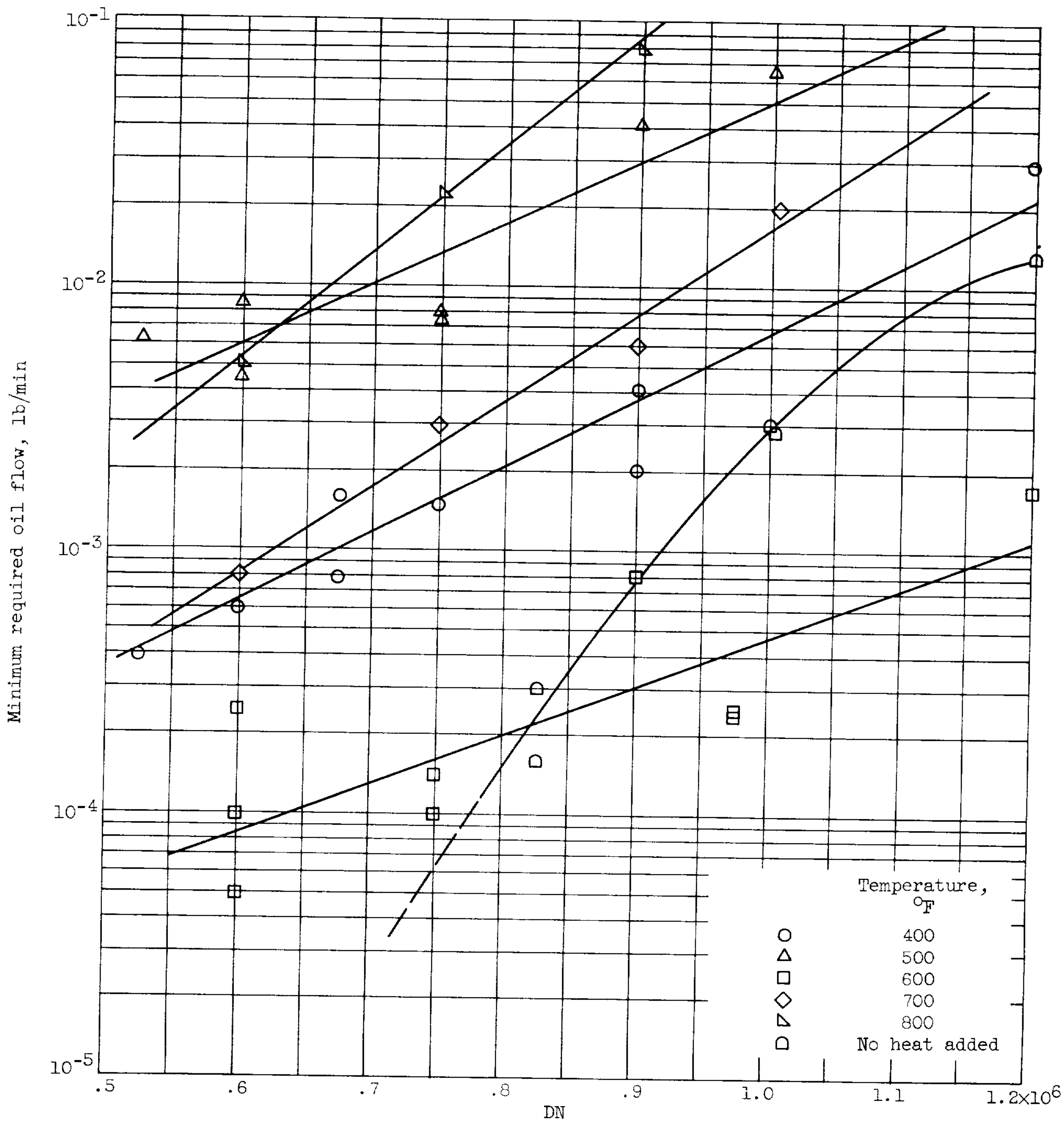
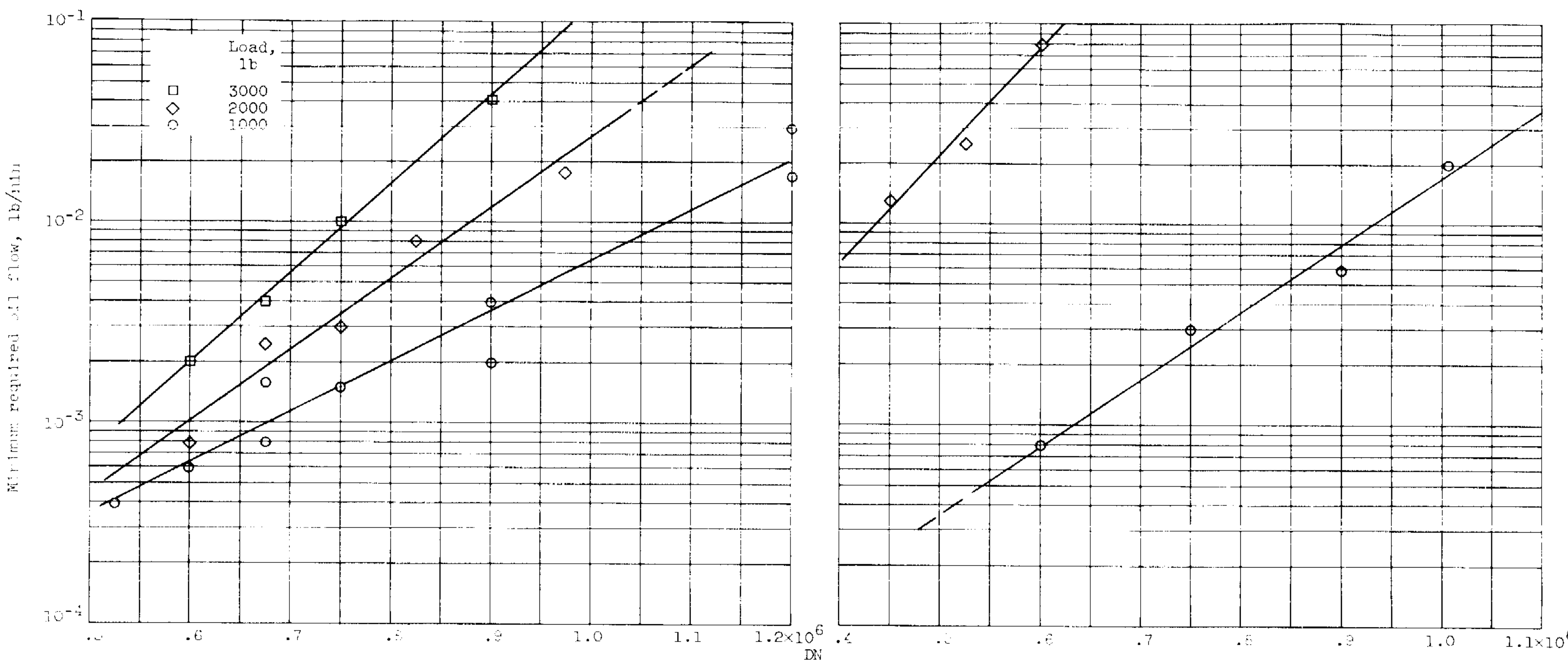


Figure 4. - Effect of DN and bearing temperature on minimum required oil flow for 1000-pound load.



(a) Bearing temperature, 400° F.

(b) Bearing temperature, 700° F.

Figure 5. - Effect of DN and bearing load on minimum required oil flow.

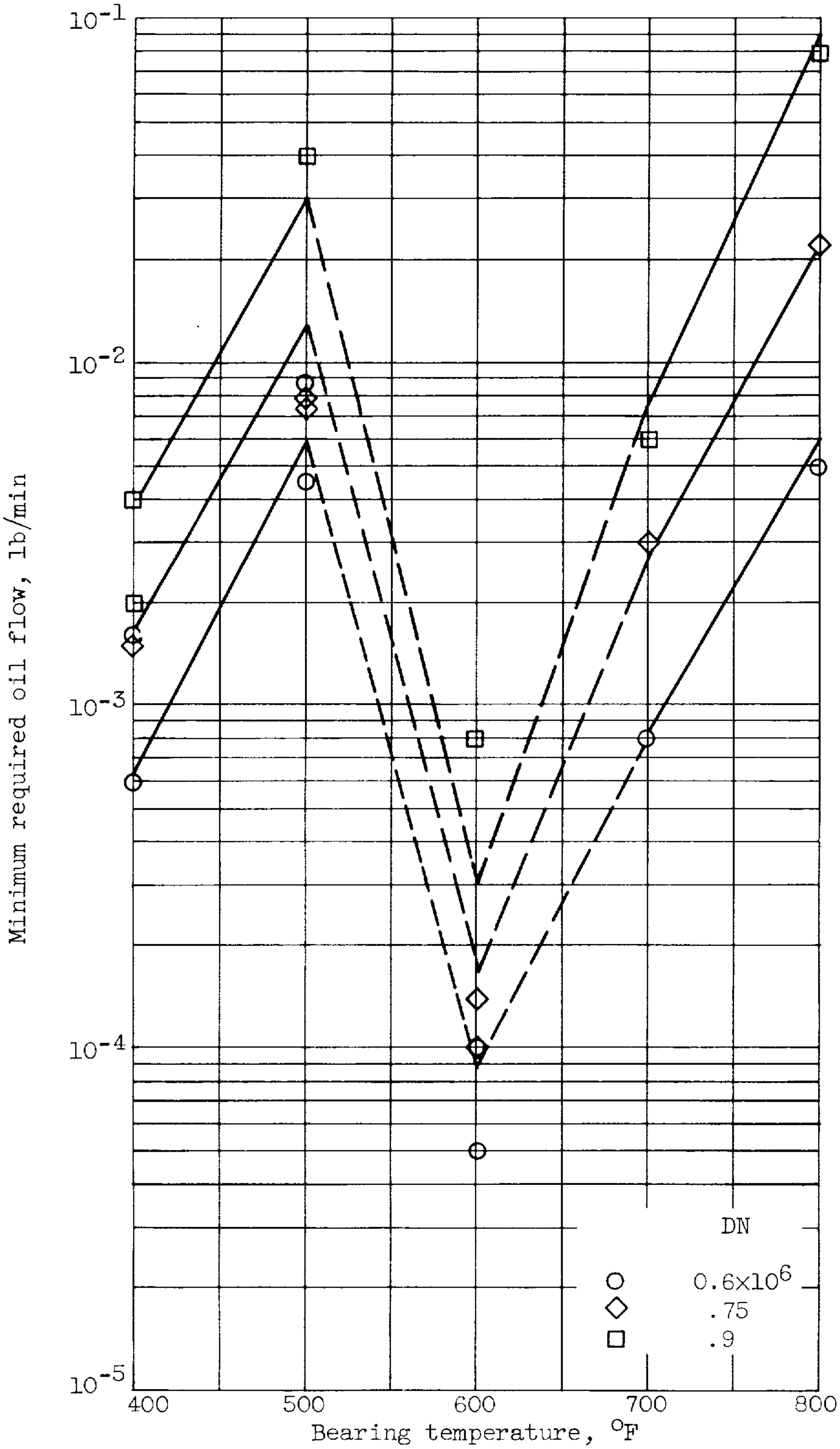


Figure 6. - Effect of bearing temperature on minimum required oil flow at three DN values. Thrust load, 1000 pounds.

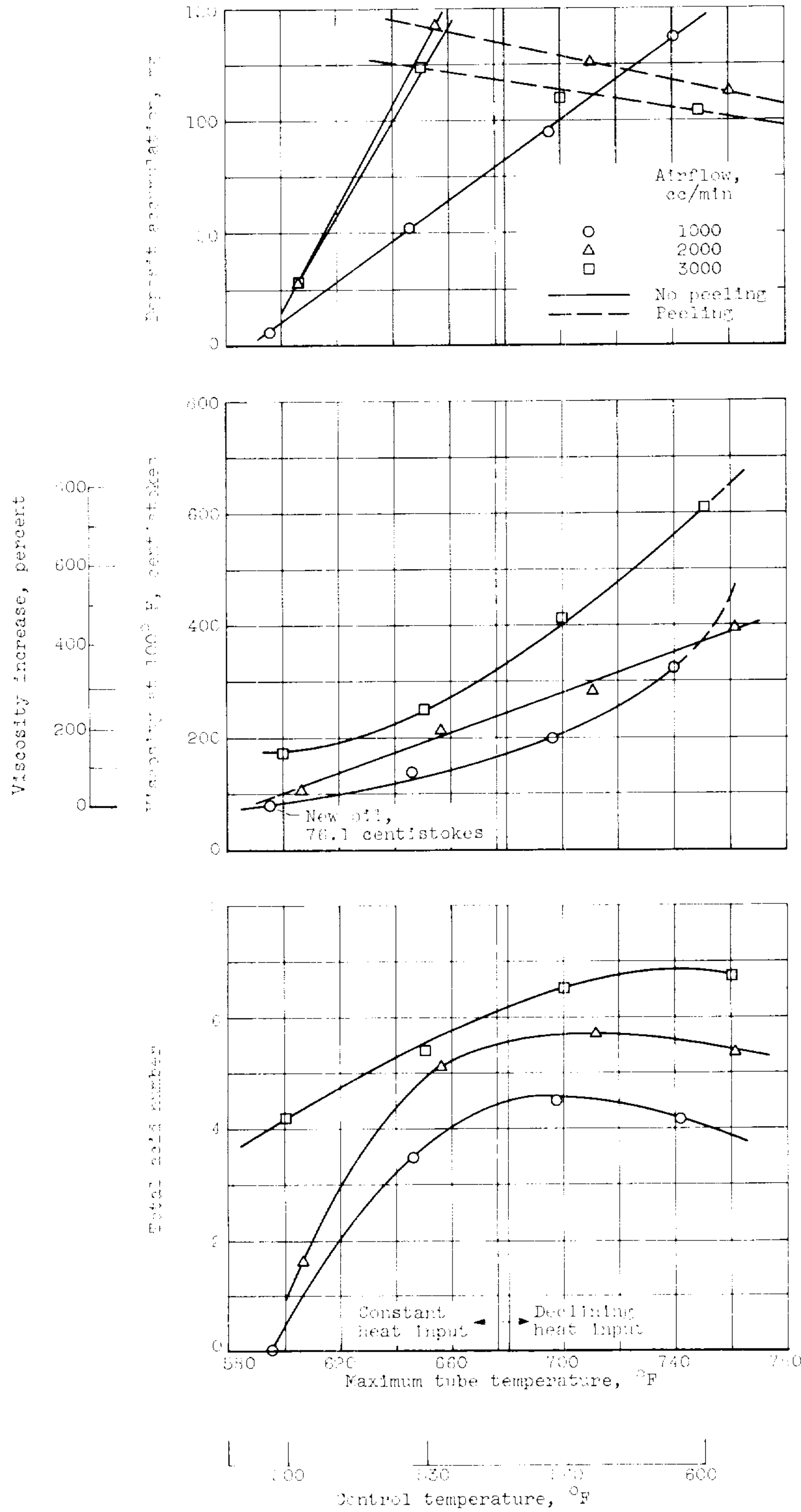
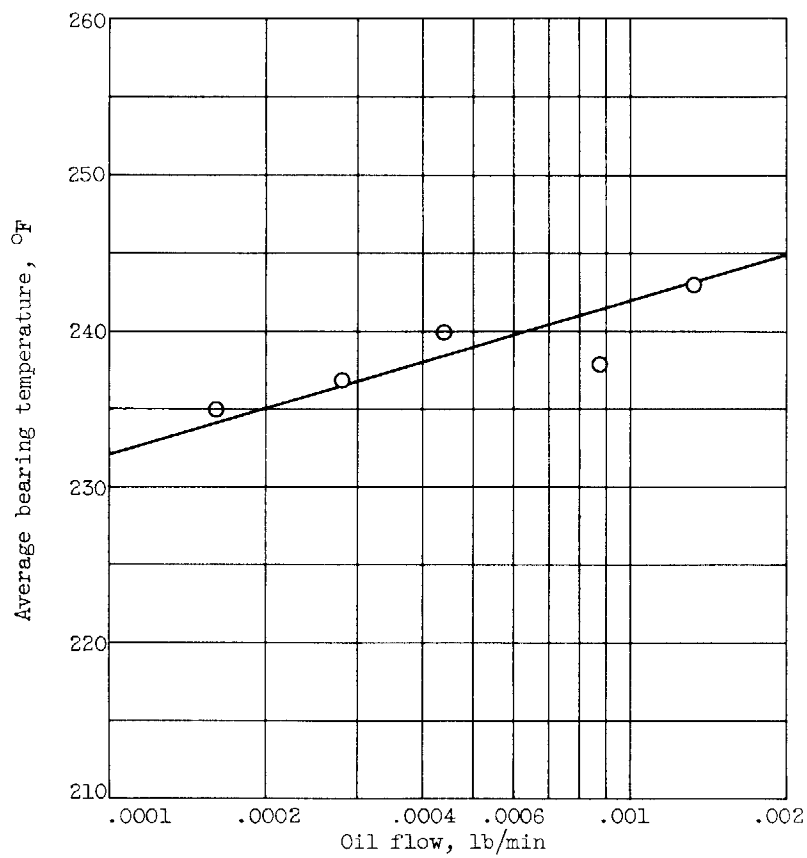
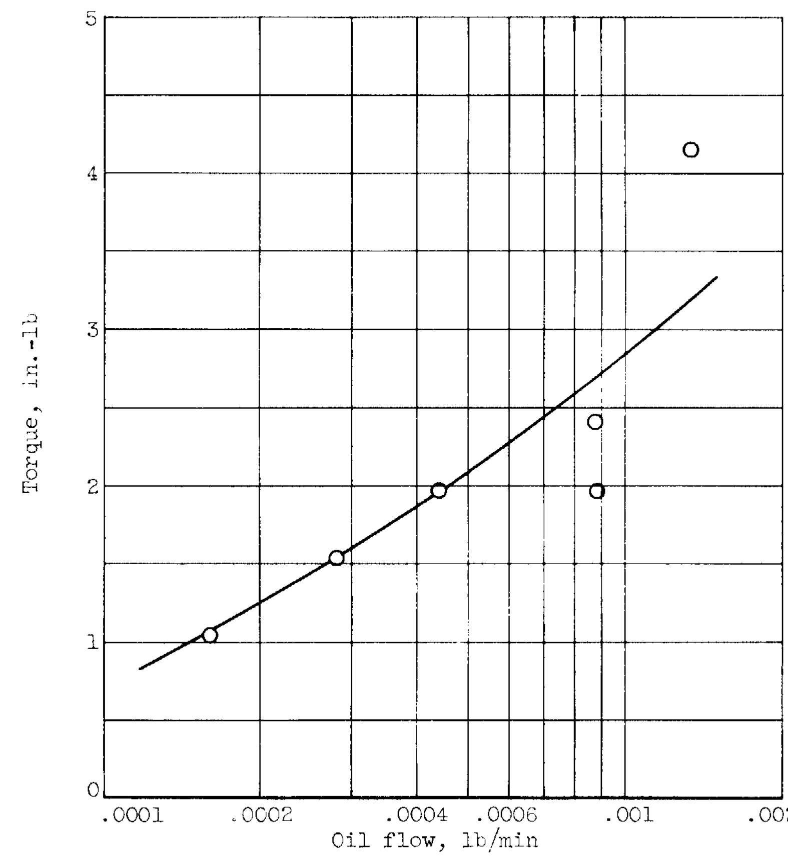


Figure 7. - Effect of temperature on acid number, viscosity, and deposit accumulation for naphthenic mineral oil, MLO 7243, in tube-type deposition tester.



(a) Bearing temperature.



(b) Bearing torque.

Figure 8. - Bearing operating temperature and torque as functions of oil flow. DN, 0.8×10^6; thrust load, 1000 pounds; no heat added.

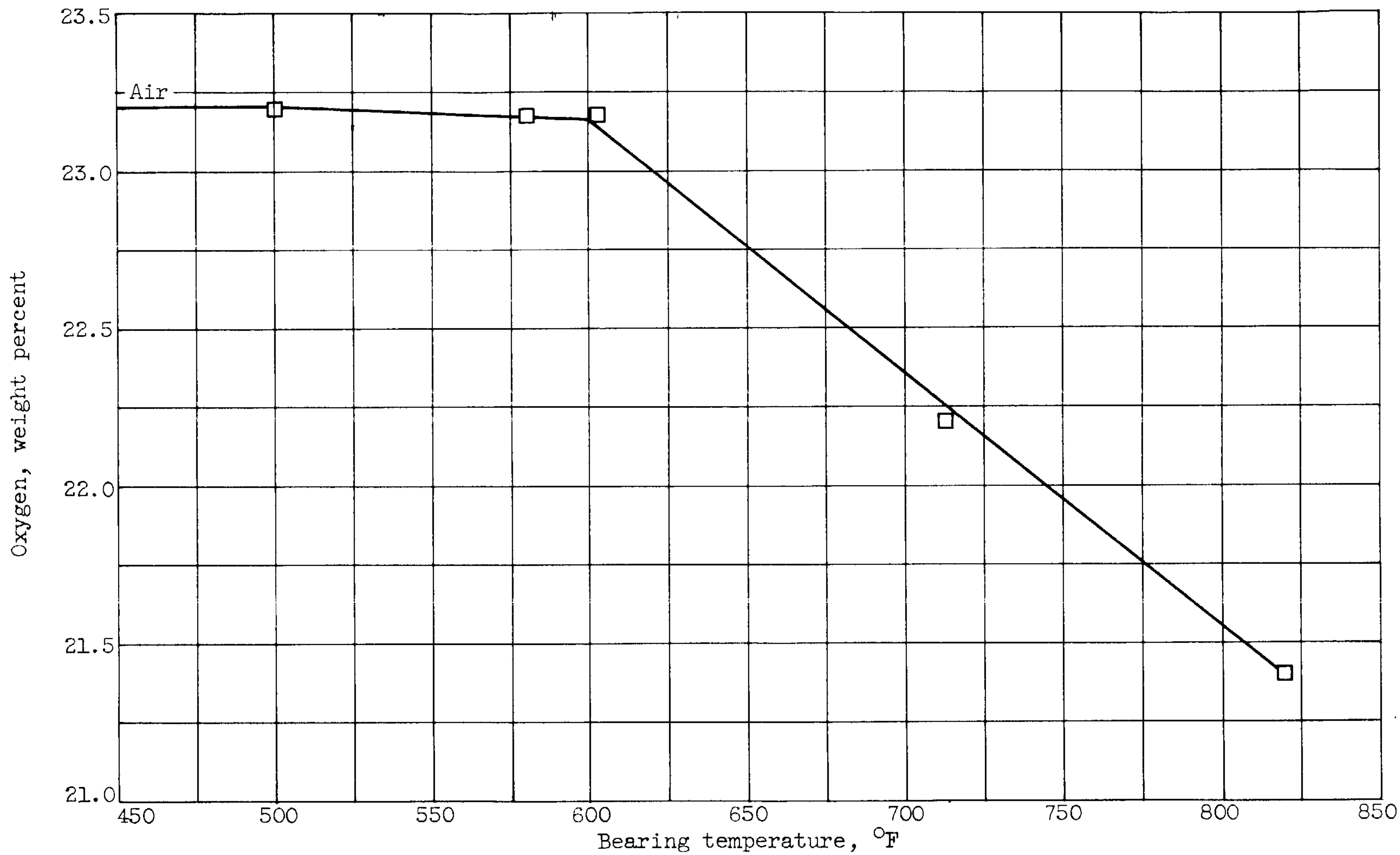


Figure 9. - Oxygen in bearing atmosphere at temperatures from 500° to 800° F. Speed, 9000 rpm; thrust load, 1000 pounds; minimum-oil-flow requirements.